PREDICTION STUDIES FOR THE PERFORMANCE OF A SINGLE CYLINDER HIGH SPEED SI LINEAR ENGINE

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Abstract

This study is prediction the performance of linear engine with spring system. To predict performance engine GT-Power software is used with small modification in friction analysis. The simulation of linear engine is done by variable speed. Performance of linear engine is determined by comparing it with conventional engine in analysis of imep, bmep, power, torque and brake specific fuel consumption. Linear engine has better performance than conventional engine.

Abstrak

Kajian ini adalah tentang prestasi enjin linear beserta sistem spring. Untuk meramal prestasi enjin tersebut, perisian GT-power digunakan dengan sedikit pengubahsuaian dalam analisis geseran. Simulasi enjin linear ini dijalankan dengan pelbagai kelajuan. Prestasi enjin linear ini ditentukan dengan membandingkan ia dengan prestasi enjin biasa dari segi analisis tentang imep, bmep, kuasa, tork dan penggunaan bahan api tentu brek.

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LIST OF SYMBOLS

Variable parameter in Friction Analysis

b = bore

D_b = bearing diameter

L_b = bearing length

N = engine speed

n_b = number of bearing

n_c = number of cylinder

P_a = atmospheric pressure

P_i = intake manifold pressure

r = compression ratio

s = stroke

U_p = piston speed

Constant in Friction Analysis

Cb = 3.03 x 10^{-4} kPa-min/rev-mm

Cg = 6.89

Cpr = 4.06 x 10^{4} kPa-mm2

Cps= 294 kPa-mm-s/m

cs = 1.22 x 10^{5} kPa-mm2

K = 2.38 x 10^{-2} s/m

K= 0.14 (spark ignition engine)

CHAPTER 1

1.1 Introduction

Most concepts of linear engines are constructed as opposed piston with complicated control devise to operate the engines. Spring has been adopted as return force of the piston movement technique. The unique of using spring as return cycle is that characteristics (stroke of the engine is not constant as conventional engine).The problem is in expansion stroke is depend on thrust force of piston. The performance of linear engine can be predicted by using GT-power software and spreadsheet. GT-power software can only simulate performance of conventional engine. However, by manipulating friction factor, the simulation can also be done for linear engine. To construct linear engine modeling, some of friction loss in conventional engine is combined into one constant fmep value and then inserted into linear engine GT-power modeling. Fmep constant value will be calculated by formulas. In order to obtain the performance of the engine, variable speed is used. Performance of linear engine that obtained from GT-power simulation is represented by graph. By compares performance of linear engine with conventional engine, the improvement of performance is known.

1.2 Problem Statement

To study and predict the performance of linear engine.

1.3 Objectives

- To study performance of linear engine by using GT-power and spreadsheet.
- To studies performance of standard conventional engine.
- To compare performance between linear and conventional engine.

CHAPTER 2

LITERATURE REVIEW

2.1 Free Piston Engine

Due to the breadth of the free piston term, many engine configurations will fall under this category. The free piston term is most commonly used to distinguish a linear engine from a rotating crankshaft engine. The piston is 'free' because its motion is not restricted by the position of a rotating crankshaft, as known from conventional engine, but only determined by the interaction between the gas and load forces acting upon it [1].

This gives the free piston engine some distinct characteristics, including (a) variable stroke and (b) the need for active control of piston motion. Other important features of the free piston engine are potential reduction in frictional losses and possibilities to optimize engine operation using the variable compression ratio [1].

2.2 Single Piston

A single piston free piston engine is shown in Fig 1. This engine is essentially consists of three parts: a combustion cylinder, a load device and a rebound device to store the energy required to compress the next cylinder charge. In the engine shown in the figure the hydraulic cylinder serves as both load and rebound device, whereas in other designs these may be two individual devices, for example an electric generator and a gas filled bounce chamber [1].

A simple design with high controllability is the main strength of the single piston design compared to the other free piston engine configurations. The rebound device may give the opportunity to accurately control the amount of energy into the compression process and thereby regulating the compression ratio and stroke length [1].

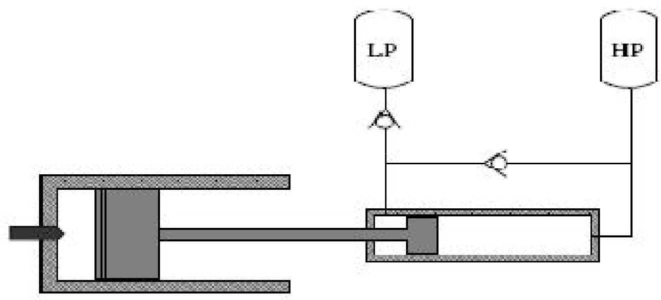


Fig. 1: Single piston hydraulic free –piston engine [1]

2.3 Free Piston Engine Features – Operating Principle.

The free piston engine is restricted to the two stroke operating principle, as a power stroke is required on every cycle. Although two stroke engines suffer from poorer performance compared four strokes, this performance gap is declining and recent years have been an increased interest in small scale two stroke engines [1].

2.4 Free Piston Engine Features – Piston Dynamics and Control

In conventional engines, the crank mechanism and flywheel serve as both piston motion control and energy storages. The piston motion control ensures sufficient compression in one end and sufficient time for scavenging in the other, while the energy storage provides energy for the compression of the next charge. In the free piston engine, the motion of the mover at any point in the cycle is determined by the sum of the forces acting upon it.

Hence, the interaction of these forces must be arranged in a way that ensures the mover motion is within acceptable limits for all types of operation if the concept is to be feasible [1].

For an engine as shown in Fig. 2a, one can derive the mover motion mathematically using a free body diagram as shown in Fig. 2b. The forces working on the mover are: combustion chamber pressure force $F_{C,}$ bounce chamber (rebound) force $F_{R,}$ load force $F_{L.}$ X denotes mover position, TDC_N and BDC_N illustrate nominal top dead centre and bottom dead centre position and ML are the mechanical limits of the motion. The mover itself will have a mass m_p [1].

Applying Newton's second law to the moving mass in Fig 2b, the piston motion can be describes with the formula below [1].

$$\sum F_i = \frac{m_p d^2 x}{dt^2} \qquad (1)$$

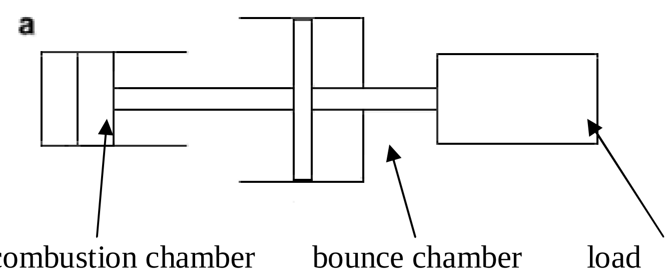


Fig. 2a: Single piston free piston engine configuration

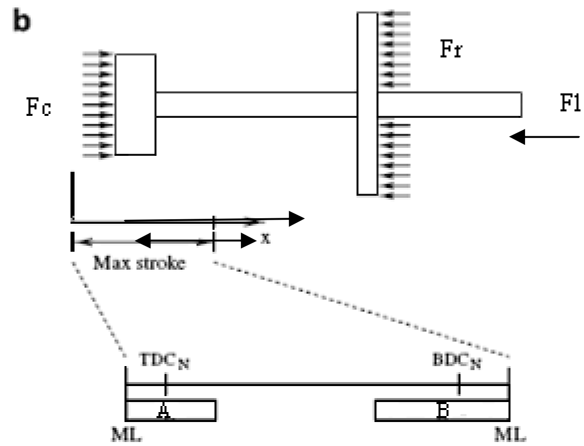


Fig. 2b: Free body diagram of the mover in a single piston free piston engine

Knowing that the combustion cylinder and the bounce chamber will have characteristics similar to those of a gas spring, it becomes clear that they will produce a bouncing type motion of the piston. Adding a load force, this must have appropriate characteristics or be subordinate the other two to ensure a reciprocating motion of the piston. If a rebound device with other force position characteristics than a bounce chamber is used, such as a hydraulic cylinder, the operational characteristics will be slightly different but the same principle will apply [1].

Fig 2b further shows the different parts of the engine stroke. Area A shows the piston position range where the compression ratio of the engine is sufficient for fuel auto ignition. For the engine to run, engine TDC must be within this area. Area B shows the piston position range where the scavenging ports are open and the burnt gases can be replaced with fresh charge. For the scavenging to be efficient, the piston needs to spend a sufficient amount of time in this area in every cycle [1].

These requirements are absolute and for the engine to be practical, an engine control system needs to be able to meet these requirements for all types of engine operation. Accurate control of piston motion currently represents one of the biggest challenges for developers of free piston engines [1].

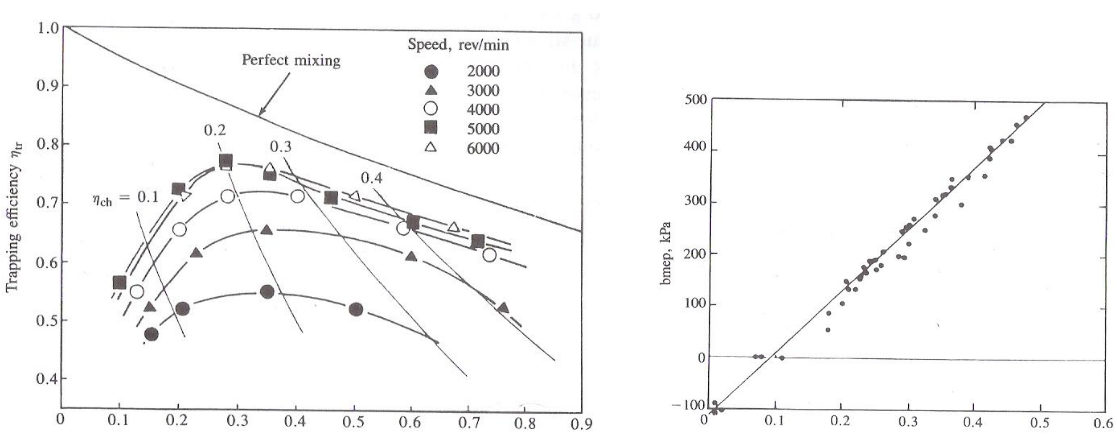
2.5 Two Stroke Cycle SI Engine.

The two stroke cycle spark ignition in its standard form employs sealed crankcase induction and compression of the fresh charge prior to charge transfer, with compression and spark ignition in the engine cylinder after charge transfer. The fresh mixture must be compressed to above exhaust system pressure, prior to entry to the cylinder, to achieve effective scavenging of the burned gases. The two stroke spark ignition engine is an especially simple and light engine concept and finds its greatest uses as a portable power source or on motorcycles where these advantages are important. Its inherent weakness is that the fresh fuel air mixture which short circuits the cylinder directly to the exhaust system during the scavenging process constitutes a significant fuel consumption penalty, and result in excessive unburned hydrocarbon emissions [2].

This section briefly discusses the performance characteristics of small crankcase compression two stroke cycle SI engines. The performance characteristics (power and torque) of these engines depend on the extent to which the displaced volume is filled with fresh mixture, i.e. the charging efficiency. The fuel consumption will depend on both the trapping efficiency. Figure 3a shows how the trapping efficiency η_{tr} varies with increasing delivery ratio Λ at several engine speeds for a two cylinder 347 cm^3 displacement motorcycle crankcase compression engine. The delivery ratio increase from about 0.1 at idle condition to 0.7 to 0.8 at the wide open throttle. Lines of constant charging efficiency η_{ch} are shown. Figure 3b shows bmep plotted against these charging efficiency values and the linear dependence on fresh charge mass retained is clear [2].

Performance curve for a three cylinder 450 cm^3 two stroke cycle minicar engine are shown in figure 4. Maximum bmep is 640 kPa at about 4000 rev/min. smaller motorcycles engine can achieve slightly higher maximum at higher speeds (7000 rev/min). Fuel consumption at the maximum bmep point is about 400 g/kW.h. Average fuel consumption is usually one-and –a-half to two times that of an equivalent four stroke cycle engine [2].

CO emissions from two stroke cycle engines vary primarily with the fuel /air equivalence ratio in a manner similar to that of four stroke cycle engines. NOx emissions are significantly lower than four stroke engines due to the high residual gas fraction resulting from the low charging efficiency. Unburned hydrocarbon emissions from carbureted two stroke engines are about five times as high as those of equivalence four stroke engines due to fresh mixture short circuiting the cylinder during scavenging. Exhaust mass hydrocarbon emissions vary approximately as $\Lambda\ (1-\eta_{tr})\ \text{Ø}$ is the fuel / air equivalence ratio [2].



delivery ratio Λ charging efficiency η_{ch}

(a) (b)

Fig. 3: a) Trapping and charging efficiencies as a function of the delivery ratio

b) Dependence of brake mean effective pressure on fresh-charge mass defined by charging efficiency

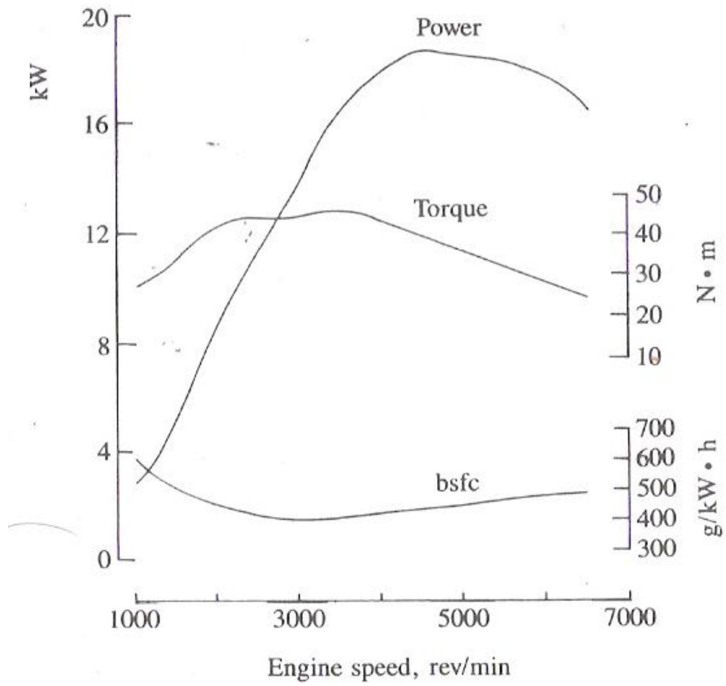


Fig. 4: Performance characteristic of a 3 cylinder 2 stroke cycle spark ignition engine

2.6 Friction

The friction forces in engine are consequence of hydrodynamic stresses in oil film and metal to metal contact. Since frictional losses are a significant fraction of the power produced in an internal combustion engine, minimization of friction has been a major consideration in engine design and operation. Engine is lubricated to reduced friction and prevents engine failure. The friction energy is eventually removed as waste heat by the engine cooling system [3].

The frictional process in an internal combustion engine can be categorized into three main components: (1) the mechanical friction (2) the pumping work (3) the accessory work. The mechanical friction includes the friction of internal moving parts such as the crankshaft, piston, rings, and valve train. The pumping work is the net work done during the intake and exhaust strokes. The accessory work is the work required for operation of accessories such as the oil pump, fuel pump, alternator and a fan [3].

We will use scaling arguments ton develop relations for the dependence of the various modes of friction work on overall engine parameters such as bore, stroke, and engine speed, then construct an overall engine friction model. The coefficients for the scaling relation are obtained from experiment data and implicitly include lubrication oil properties such as viscosity [3].

The following are the mechanical components friction for internal friction engine which are going to use in analyzing single piston free piston linear engine.

a) crankshaft-main bearings
b) crankshaft-seal
c) piston-rings
d) piston-gas pressure

a) Crankshaft-main bearings [3].

The friction mean effective pressure of a journal array with η_b bearings such as the crankshaft main bearings or the connecting rod bearings scales linearly with engine speed, assuming constant bearings clearance and oil viscosity.

$$fmep_{bearings} = \frac{c_d n_b N D_b^3 L_b}{n_c b^2 s} \qquad (2)$$

b) crankshaft-seals [3].

The crankshaft bearings seals operate in a boundary lubrication regime, since the seals directly contact the crankshaft surface. As the normal force, which the seal lip load, is constant, the friction force will be constant and the friction mean effective pressure of the crankshaft bearing seal will be independent of engine speed, and will scales as

$$fmep_{seals} = \frac{c_s D_b}{n_c b^2 s} \qquad (3)$$

Patton et al (1989) suggest a proportionality constant Cs = 1.22 x 10^5kPa –mm^2. If the bearing is not sealed, oil will leak out at the ends, so oil is pumped at relatively low pressures through internal passages to the bearings annulus.

c) piston-rings and piston-gas pressure [3].

The friction of the piston and rings results from contact between the piston skirt and the ring pack with the cylinder bore. The cylinder bore is rougher than a journal bearing bore since the cylinder bore must retain some oil during operation. The ring seals the combustion chamber, control the lubrication oil flow and transfer heat from the piston to the cylinder. In order to preserve a seal against the cylinder bore, each ring has some amount of radial tension.

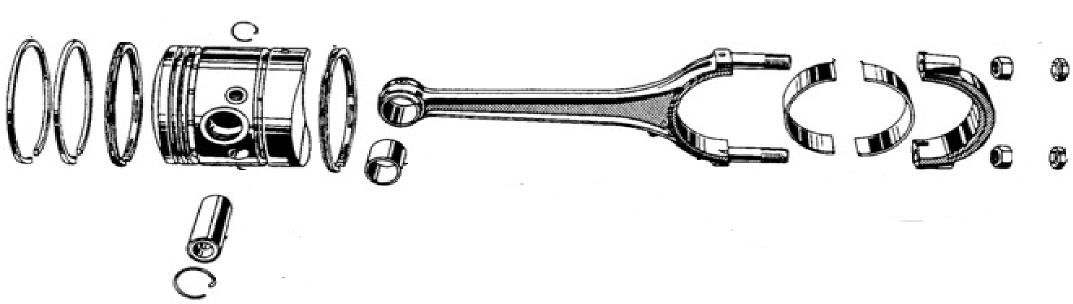


Fig. 5: Examples of piston assembly

The friction force of the piston rings has two components, one resulting from the ring tension and the other component from the gas pressure loading. The component of piston friction due to ring tension in the mixed lubrication regime will have a friction coefficient inversely proportional to the engine speed. The piston ring fmep scaling is

$$fmep_{rings} = \frac{c_{pr}}{b^2}\left(1 + \frac{1000}{N}\right) \qquad (4)$$

A correlation for the component of piston friction due to the gas pressure loading recommended by Bishop (1964) is

$$fmep_{gas\ load} = \frac{c_g P_i}{P_a}\left(0.088r + 0.182r^{(1.33-KU_P)}\right) \qquad (5)$$

2.7 Simulation of Two Stroke Compression Ignition Hydraulic Free Piston

Engine

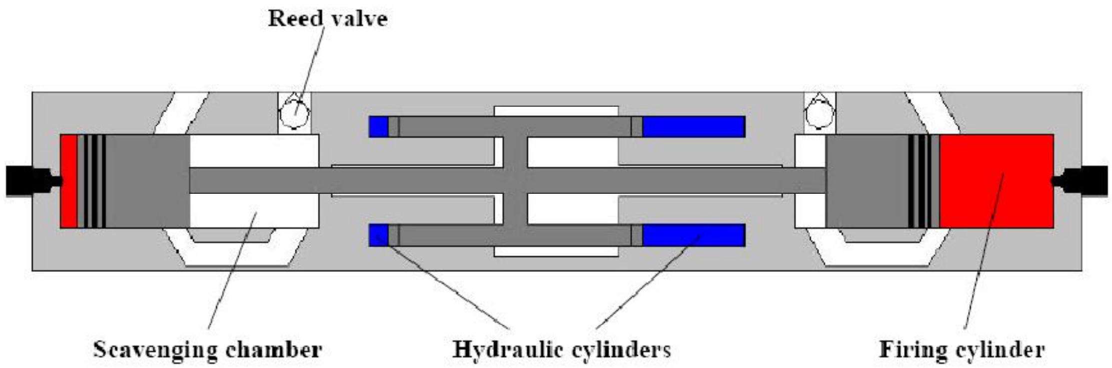


Fig. 6: The free piston engine [4]

Table 1: Free piston engine specification [4]

Bore	90 mm
Stroke	112~114 mm
Output	13-18 kW
Cycle Frequency	1700 1/min (rpm)
Outer dimension	1100 x 350 x 200 mm
Weight approximately	120 kg
Common rail fuel injection up to 1350 bar pressure	
Direct injection	

Piston Motion of the Free Piston Engine [4].

- The piston does not follow the regular crank and connecting rod motion.
- The piston motion is not symmetric around TDC/BDC.
- The piston leaves the dead center at a high speed than it approaches them.

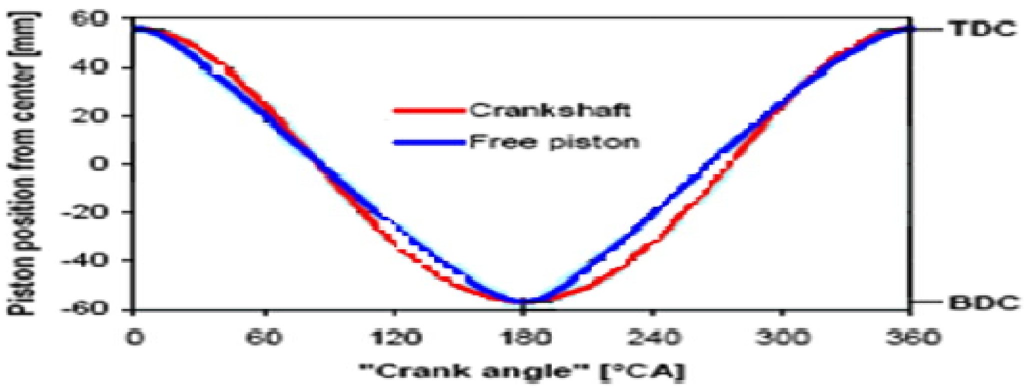


Fig. 7: The piston position against crank angle

Using Piston Motion of the FPE in GT-POWER [4].

- Piston motion object: EngCylGeomUser
- Piston motion as an XY Table.
- Piston motion either measured or simulated.
- The reference plane is at BDC.
- The XY Table starts at TDC.

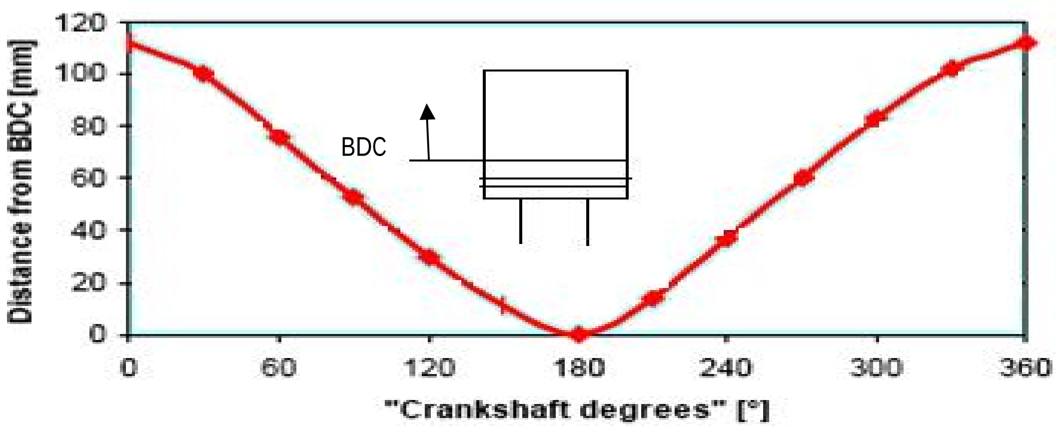


Fig. 8: Effect of distance from BDC against crankshaft degrees.

Using Piston Motion of the FPE in GT-Power [4].

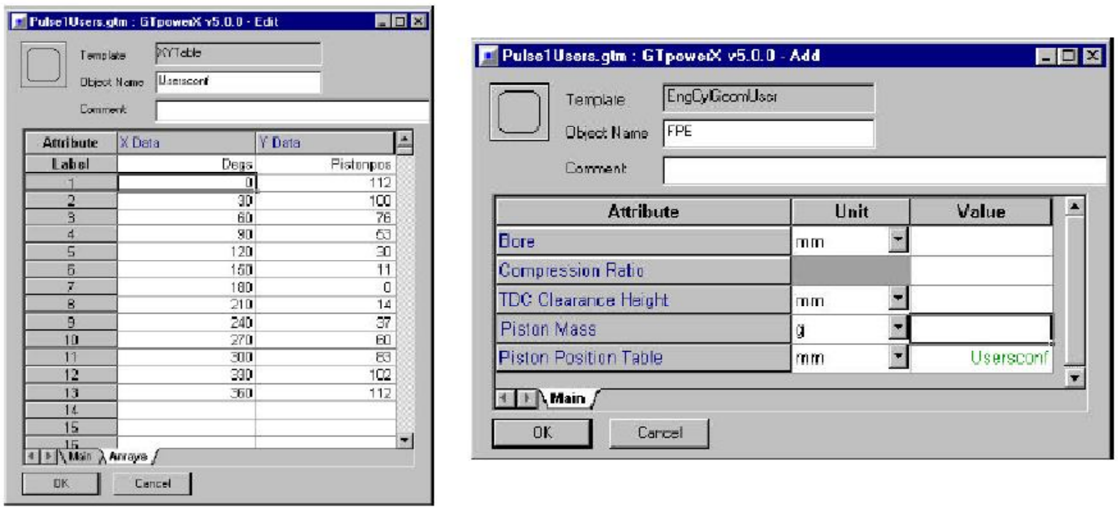


Fig. 9: Figure of piston adjustment in GT-power.

Input piston position curve: 706 points

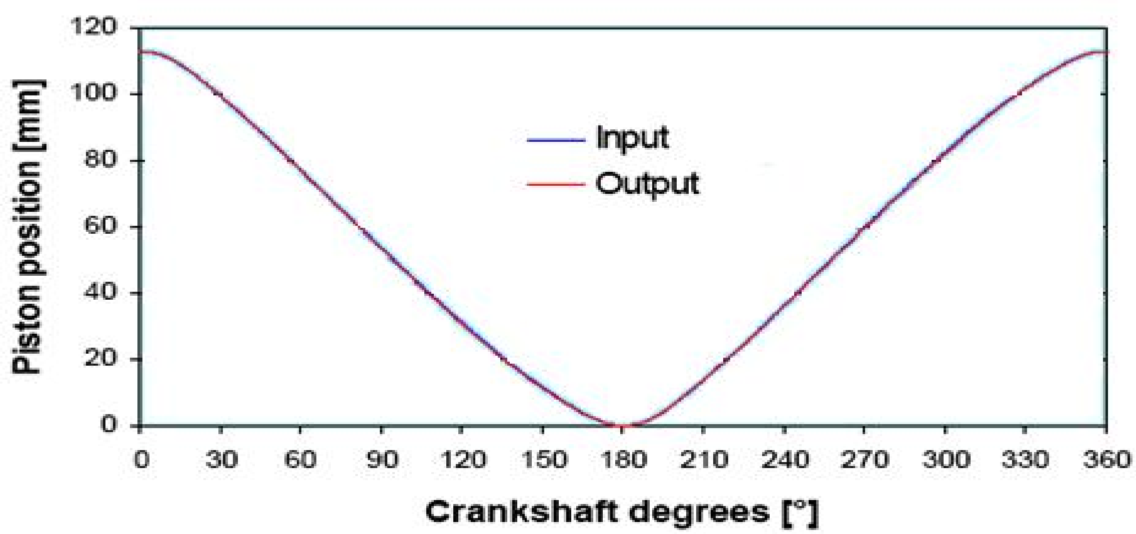


Fig. 10: Piston position with variable crankshaft degrees